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RESEARCH MEMORANDUM

VIBRATION OF LOOSELY MOUNTED TURBINE BLADES DURING SERVICE OPERATION OF A TURBOJET ENGINE WITH CENTRIFUGAL COMPRESSOR AND STRAIGHT-FLOW COMBUSTION CHAMBERS

By W. C. Morgan, R. H. Kemp
and S. S. Manson

Lewis Flight Propulsion Laboratory
Cleveland, Ohio

NATIONAL ADVISORY COMMITTEE
FOR AERONAUTICS

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NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

RESEARCH MEMORANDUM

VIBRATION OF LOOSELY MOUNTED TURBINE BLADES DURING SERVICE OPERATION OF A TURBOJET ENGINE WITH CENTRIFUGAL COMPRESSOR AND STRAIGHT-FLOW COMBUSTION CHAMBERS

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SUMMARY

An experimental investigation was conducted to determine the vibration characteristics of loosely mounted turbine blades during service operation of a turbojet engine. High-temperature strain gages were used to measure the turbine-blade vibrations.

The turbine blades studied were 4 inches long; the experimental investigation was confined to blades having 0.03-inch and 0.06-inch amplitude of tip movement at room temperature in the plane of the turbine wheel. Vibration during service operation occurred in the first bending and first torsional modes of the turbine blades. In addition, a small number of complex modes of vibration was observed. Sources of vibration excitation were present at frequencies related to the number of nozzle blades and combustion chambers and to multiples of the first-order turbine speed.

A comparison was made between the vibrations of the loosely mounted blades and those observed during a previous investigation of similar blades tightly mounted in a turbine wheel. For the particular engine used, in which the vibratory-stress levels were generally low, the comparison did not show the existence of any appreciable vibration damping attributable to looseness in turbine-blade mounts. The loosely mounted blades appeared to be more susceptible to vibration in the first bending mode. The possibility exists, however, that in the presence of higher excitation forces, a relative lowering of vibratory-stress levels might result from loose blade mounting.

INTRODUCTION

A recent modification in turbine-blade fastening has been that of increasing the clearance between blade-base serrations and the corresponding serrations in the rim of the turbine wheel. Several turbojet engines have been fabricated with loosely mounted turbine blades. The use of loose blades may possibly increase the vibration damping inherent in inserted-blade design.

The amplitude of blade-tip movement in the plane of the turbine wheel varies among the several existing turbine-blade designs and among the blades of any individual turbine wheel because of manufacturing tolerance limits. For example, one manufacturer specifies that the tip amplitude shall be within the limits 0.002 to 0.060 inch for a blade 4 inches long. The factor of blade length must be considered when the degree of looseness in blade-base fit is specified by amplitude of blade-tip movement.

The problem of turbine-blade vibration has hindered the development of the turbojet engine (reference 1). A study of some recent service failures in turbine blades, however, has indicated that the failures might be attributed to vibration fatigue (reference 2).

A British investigation has been made of certain blade-vibration problems encountered during development of gas turbines and compressors (reference 3). In this study of the damping characteristics of turbine blades, approximately 70 percent of the damping was attributed to the blade-root attachment in the wheel; the remainder was shared by aerodynamic and material damping. These values apply only to turbine blades retained in serrated dovetails. Another conclusion drawn from the results of the British investigation was that a turbine-blade root could be considered as tightly mounted at service operation turbine speeds, irrespective of the root tightness when stationary. This conclusion is at variance with the concept that looseness of a turbine blade in the wheel rim will effect a considerable increase in damping of the turbine-blade vibration.

A program is in progress at the NACA Lewis laboratory to investigate the problems associated with vibration in turbine blades. As a part of this general program, an investigation was made to determine the vibration characteristics of loosely mounted turbine blades during actual service operation of a turbojet engine. The measurements of vibration frequency and strain were obtained from high-temperature strain gages mounted on the turbine blades. The tip amplitudes approximated the average and maximum movement specified by the manufacturer for the type of turbine blade employed in the experimental investigation.

A comparison was made between the results from a study of vibration characteristics of tightly mounted blades (reference 4) and those obtained during the investigation of loosely mounted turbine blades. The comparison was based on the effects produced by loose blade-base mounting on natural frequency, response to excitation forces, and damping of vibration in the turbine blades.

In the analysis of the results, attention was given to the magnitude of the excitation forces. An investigation has been made of the effects produced by varying the magnitude of an excitation force applied to a cantilever blade fastened in a rotor in such a manner as to have a considerable amount of mechanical damping in the base mounting (reference 5). The damping of this type of blade mounting was affected to a considerable extent by the magnitude of the exciting force. It is therefore probable that this factor is important in interpretation of the data obtained from the turbine blades investigated, in which the vibratory-stress level was generally low.

EXPERIMENTAL EQUIPMENT AND PROCEDURE

The turbojet engine used in this investigation is described in reference 4. The engine is a straight-flow type with a centrifugal compressor, 14 combustion chambers, 48 nozzle vanes, and a single-stage turbine. The turbine blades are unshrouded and are attached in the rim of the turbine wheel with serrated dovetails.

Certain modifications were made to the turbojet engine in order to provide a passage for the lead wires from the strain gages mounted on the turbine blades to the slip-ring unit at the forward end of the engine. Axial holes were bored through the turbine wheel and shaft, the compressor, and the several shafts connecting these components. A hollow auxiliary shaft provided a connection between the forward hub of the compressor and the slip-ring unit mounted on the accessory housing.

Modification of the bases of the turbine blades used in the experimental investigation was necessary in order to obtain the desired amount of looseness in fit between the blade bases and the serrations in the turbine-wheel rim. The turbine blades were 4 inches long, and the bases were modified to provide two degrees of looseness. One blade had an amplitude of tip movement in the plane of the wheel of 0.03 inch; a similar blade had an amplitude of movement at the tip of 0.06 inch. These amplitudes were selected as approximately representative of average and maximum looseness for turbine blades of the type used in the investigation.

The turbine blades were originally intended to be mounted tightly and then positioned positively with respect to axial position in the turbine wheel by peening the blade-base dovetail. The removal of sufficient material from the blade-base serrations to provide increased clearance in the attachment made it necessary to employ a different method of retaining the blade in the wheel. The principal requirement of the fastening was positive positioning with minimum restriction to movement of the turbine blade in the plane of the wheel and within the limits of the desired blade-tip amplitude. It was also considered necessary that the method of attachment permit removal and replacement of experimental blades without a major disassembly of the engine. The modifications made to the turbine-wheel rim and to the dovetail sections of the experimental turbine blades and the blade-base modification made to accommodate strain-gage lead wires are shown in figure 1.

When the turbine blades were assembled in the wheel, a positioning guide and a hardened-steel sphere were placed in that part of the wheel passage having the larger diameter; a modified blade was then inserted in the wheel dovetail and the adjustment screw tightened until the sphere was seated in the hemispherical cavity machined in the blade dovetail. The adjustment screw was then loosened until there was no appreciable restraint to blade movement in the plane of the wheel. When the adjustment was satisfactory, the screw was locked in position.

High-temperature resistance-wire strain gages were used to obtain data from the loosely mounted turbine blades during service operation of the engine. The construction and the mounting of the strain gages were similar to those of the multiple-loop type, described in references 4 and 6, with some improvements that have resulted from continued research on high-temperature strain gages. The strain-sensitive wire was a platinum-iridium alloy; Sauereisen No. 32 cement was used as the mounting material. A precoat of a high-temperature ceramic was fired on the surface of an experimental blade before the strain gage was mounted. This ceramic coating, L-6AC, is a development of the National Bureau of Standards. After the strain gage had been baked on a turbine blade at low temperature, the blade was placed in a high-temperature oven and heated sufficiently to stabilize the strain-sensitive characteristics of the wire and to complete the bonding of the cement. The strain gages were applied near the blade bases along the trailing edges on the convex side.

A photograph (fig. 2) of one of the instrumented turbine blades shows the precoat, the location of the high-temperature strain gage, and the ceramic-lined Inconel conduit enclosing the strain-gage lead wires. The turbine blades were inserted and fastened in the wheel by the method previously described, and the lead-wire conduit was attached to the surface of the turbine wheel by small metal straps spot-welded to the wheel. The lead wires were then attached to a terminal plate at the center of the turbine wheel.

Lead wires were run from the terminal plate, through the engine, and attached to the slip-ring rotor on which were mounted the inactive arms of the Wheatstone bridges. Brushes bearing on the slip rings carried current to the strain-gage bridges and transmitted the bridge output signal to the stationary instrumentation in the control room.

The experimental procedure consisted in operating the engine in a pendulum-type, sea-level test stand over the entire range of turbine speeds at exhaust temperatures similar to those that occur during regular service operation. The speed ranged from idling (4000 rpm) to full turbine speed (11,500 rpm). As the speed was slowly increased, the strain-gage signals were under constant observation. At the appearance of signals indicative of vibration, the turbine speed was held constant and the signals were recorded. These oscillograph records were used in the computation of turbine speeds, vibration frequency, and vibratory stress. (Reference 4 describes details of the methods.)

RESULTS AND DISCUSSION

Two oscillograph records representative of the data obtained during engine operation are given in figure 3. These records show strain-gage signals indicating turbine-blade vibration at nominal engine speeds of 10,000 and 11,500 rpm. In each of the records shown, the uppermost oscilloscope trace was produced by a 400-cycle-per-second signal from a vacuum-tube tuning fork. This trace served as a frequency standard for accurate determination of turbine speed and vibration frequency. The two oscilloscope traces below the frequency standard show the variation in the output of the two strain-gage bridges, one arm of each being a high-temperature strain gage. The remaining signal was produced by an impulse tachometer mounted on the turbojet engine. Two impulses occurred during each revolution of the turbine wheel.

In figure 3(a), the upper strain-gage trace shows the presence of vibration in a turbine blade; the other turbine blade was not vibrating and the signal shown in the lower strain-gage trace is therefore indicative only of the level of electrical interference caused by small variations in resistance between the slip rings and the brushes.

A vibration signal under similar conditions is presented in figure 3(b) in which the second strain-gage signal has been short-circuited in order to show the absence of electrical interference in the instrumentation used for amplification, observation, and recording of strain-gage signals.

Vibration was observed in the turbine blade with 0.03-inch freedom of tip movement and also in the blade having 0.06-inch amplitude at the tip. An analysis of the data obtained from the turbine blade with the 0.03-inch amplitude of tip movement is presented in figure 4. The symbols indicate the occurrence of vibration observed during the investigation. The order lines indicate the loci of points of which the frequency is a definite multiple of the turbine speed. These lines show the frequency of any exciting force that can occur at the definite multiples of turbine speed.

For example, the turbojet engine used in the investigation has 14 combustion chambers. From consideration of the geometry of the engine, it is therefore possible that excitation forces exist at a fourteenth multiple of the turbine speed. The critical-speed diagram (fig. 4) shows that vibration was observed at orders of excitation, which could be attributed to interruption of the gas forces by the combustion chambers and nozzle vanes, and at several orders of rotative speed, probably because of inequalities in mass flow.

The turbine blade vibrated in the first bending mode at approximately 1150 cycles per second and in the first torsional mode at about 1900 cycles per second. These mode frequencies were determined during previous bench tests of turbine blades. Complex modes of vibration at higher frequencies were also excited, apparently by the higher orders of intermittent gas loading. Only one of these complex-mode vibrations occurred within the service-cruising speed range, approximately 9600 to 11,500 rpm.

The critical speeds and frequencies obtained from operation of a turbine blade with 0.06-inch amplitude of tip movement are presented in figure 5 in the same manner as in figure 4. The turbine blade vibrated in the first bending mode at approximately 1150 cycles per second and in the first torsional mode at about 1800 cycles per second. Only one complex-mode vibration was of appreciable magnitude; the excitation-force frequency coincided with the twenty-eighth order of turbine speed, which is also the second order of the 14-combustion chambers. No significant differences in vibratory-stress levels were observed between this turbine blade and the blade with the lesser amplitude of tip movement.

The results of a similar investigation of tightly mounted blades of the same design (reference 4) are shown in figure 6. Comparison between these results and those obtained with loosely mounted blades indicates that the vibration characteristics were nearly similar, both as to frequency and stress level.

On the basis of this comparison, any gain in damping effect produced by loose mounting appeared negligible. The vibration characteristics of the loosely mounted turbine blades differed from those of tightly mounted blades in one respect: the loosely mounted blades showed a tendency toward vibration of brief duration in first bending mode at many different turbine speeds in addition to those speeds coincident with the occurrence of a sustained vibration. The tightly mounted blades vibrated perceptibly only at definite turbine speeds coincident with the condition of sustained vibration associated with resonance.

Furthermore, when vibration of appreciable magnitude occurred in the case of the loosely mounted blades, the vibration was sustained at a nearly constant value over a small but definite range of turbine speed. A condition of vibration in tightly mounted blades occurred only within a much more limited range of turbine speed. It was also observed that the loosely mounted blades would sometimes change momentarily from a complex-mode vibration to a first bending-mode vibration.

Several factors must be considered in the analysis of the results. With the exception of the two experimental blades, all turbine blades were tightly mounted. This factor may have caused a relatively greater amount of compressive effect in the rim at the location of the loosely mounted blades than would have occurred in a turbine wheel completely fitted with loosely mounted blades. The differences in vibration characteristics observed for the two types of mounting, however, warrant the assumption that some degree of looseness existed in the experimental blade mounts during engine operation.

Another possible effect of the presence of tightly mounted turbine blades in the wheel is the transmission of vibration from a blade at resonance to other blades in a remote location, either through the base common to both, or in the manner exemplified by excitation of one tuning fork by another of the same natural frequency. Previous examination of a considerable number of turbine blades has shown, however, that natural frequency varies considerably among similar turbine blades. In addition, the method of dovetail insertion would tend to prevent any such vibration because of its inherent damping and because of its tendency to change the effective length of a turbine blade, dependent on the individual fit of the base mount in the wheel dovetail.

The vibratory-stress levels were not excessively high in either the loosely mounted or tightly mounted blades. As a corollary, the excitation forces may be assumed to be low. The results of a recent investigation have shown that the magnitude of the excitation force producing a vibration has an important bearing on whether the level of vibratory stress is affected by looseness of blade mounting (reference 5). Some of the data obtained during that investigation are presented in figure 7. The blade was a cantilever type, with a base mount similar in design to that employed in axial-flow-compressor blades. The data presented are all for the first bending mode.

In figure 7, the stress in the tightly mounted blade is proportional to the excitation force regardless of turbine speed. For the loosely mounted blade, the stress is dependent on both turbine speed and exciting force. At 7950 rpm, for example, the blade is effectively tight until the excitation force is greater than 1 pound. Greater excitation forces apparently produce some rubbing action in the mount and the stress for a given excitation force is therefore lower than in the case of the tightly mounted blade. At a speed of 10,150 rpm, the centrifugal force is sufficient to hold the blade tight until the excitation force approaches 2 pounds; thereafter, the stress for a given excitation force is lower than that induced in the tight blade but higher than the stress that occurred in the blade with the same degree of looseness when operated at a lower rotor speed.

The additional damping inherent in a loose blade mount may not become effective unless the exciting force is of sufficient magnitude to offset the action of centrifugal force. A loose mounting would conceivably be of value as a vibration damper.

SUMMARY OF RESULTS

High-temperature resistance-wire strain gages were used to determine the vibration characteristics of loosely mounted turbine blades. The experimental blades were excited into vibration by forces present in a turbojet engine during service operation. The blades were 4 inches long; one had a tip amplitude of 0.03 inch in the plane of the turbine wheel; the other had a tip amplitude of 0.06 inch. For both blades, the majority of the vibrations observed were in first bending and first torsional modes. Some complex-mode

Abstract

Vibration characteristics of loosely mounted turbine blades were determined during service operation of a turbojet engine. High-temperature strain gages were used to measure turbine-blade vibrations. Vibration occurred in first bending and first torsional modes; in addition, a small number of complex-mode vibrations was observed.

Comparison was made between vibrations in loosely mounted blades and those observed during a previous investigation of similar blades tightly mounted in a turbine wheel. The comparison did not indicate that any considerable gain in damping was effected by the use of loosely mounted blades.

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Vibration of Loosely Mounted Turbine Blades during Service Operation of a Turbojet Engine with Centrifugal Compressor and Straight-Flow Combustion Chambers.

By W. C. Morgan, R. H. Kemp, and S. S. Manson

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November 1949

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vibrations were also observed. The sources of excitation were of frequencies related to the number of nozzle blades and combustion chambers and to multiples of the first-order turbine speed.

Comparison was made between the results of this investigation and those obtained during similar research on tightly mounted turbine blades. The results indicated that loose blade mounting had little effect on frequency of vibration and vibratory-stress levels. The loosely mounted blades were observed to be more susceptible to vibration in the first bending mode.

The results presented were obtained from vibration in turbine blades that were subjected to low excitation forces; in the presence of more severe excitation forces, however, the increased damping attributable to loose mounting of a turbine blade may be a factor of importance in reducing the level of vibratory stress.

Lewis Flight Propulsion Laboratory,
National Advisory Committee for Aeronautics,
Cleveland, Ohio.

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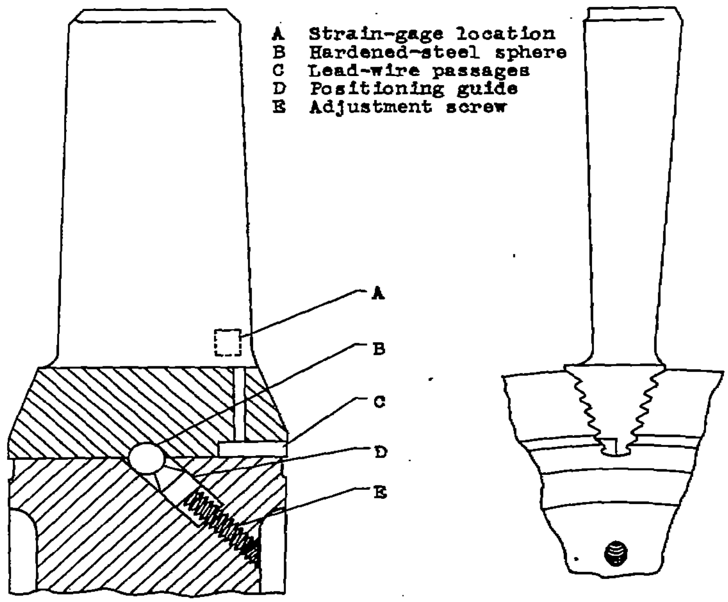


Figure 1. - Method of blade-root fastening used in investigation of vibration in loosely mounted turbine blades.

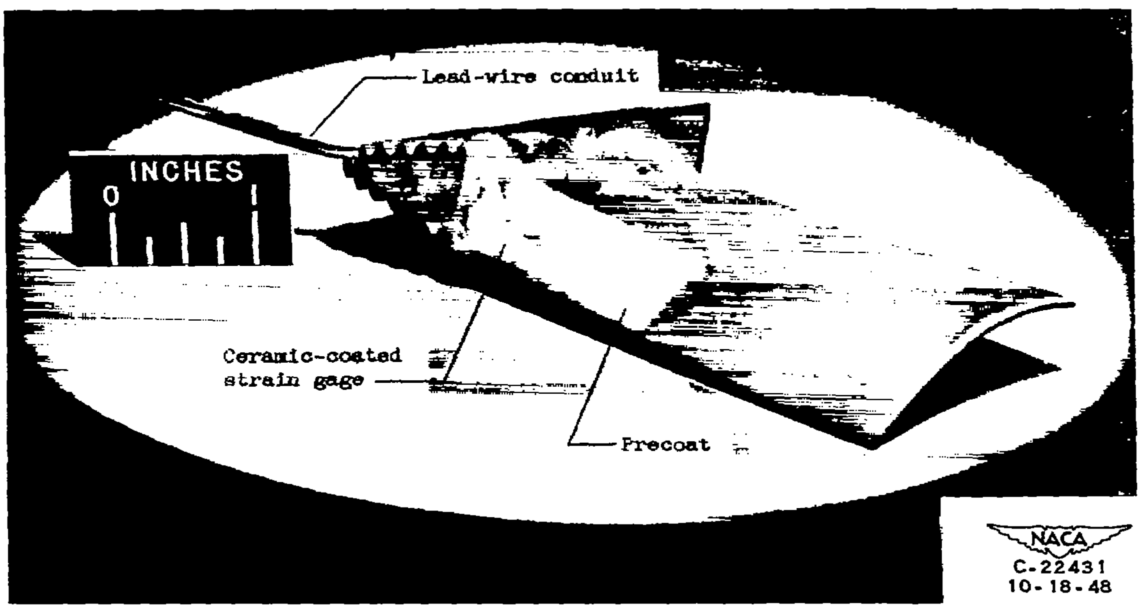
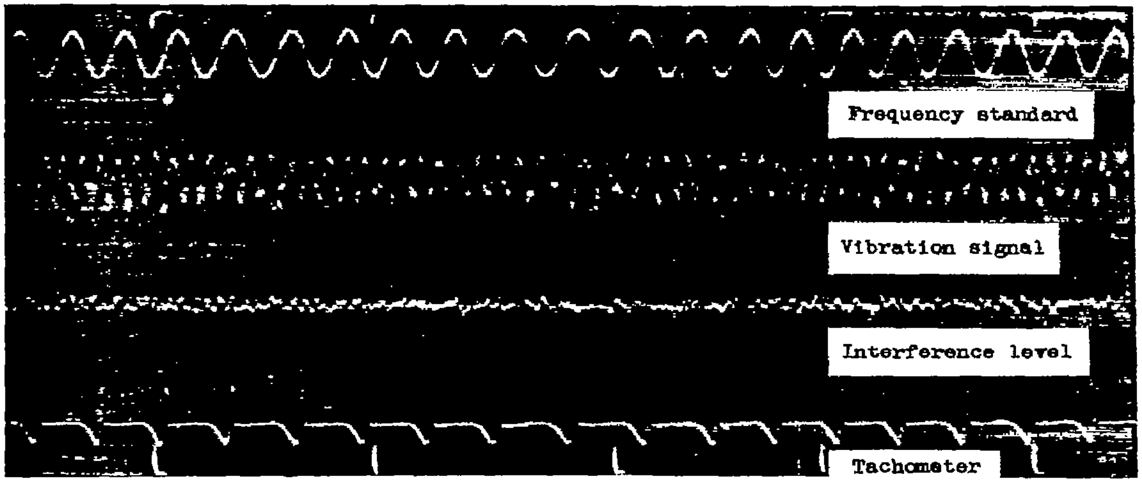
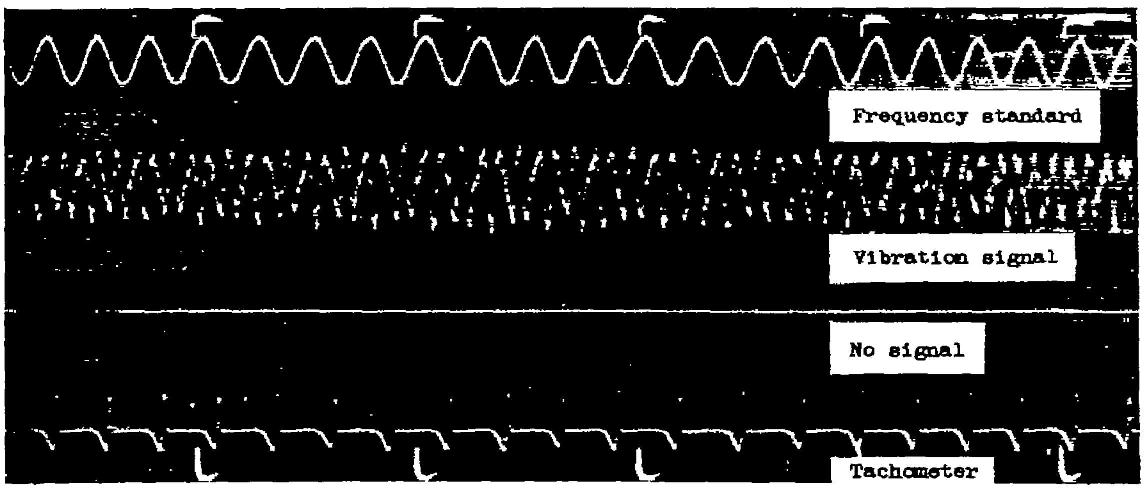


Figure 2. - High-temperature strain-gage instrumentation of turbine blade showing location of strain gage and lead-wire conduit.



(a) Engine speed, 10,000 rpm.



(b) Engine speed, 11,500 rpm.

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Figure 3. - Oscillograph records of data used for computing turbine speed, vibration frequency, and vibratory stress.

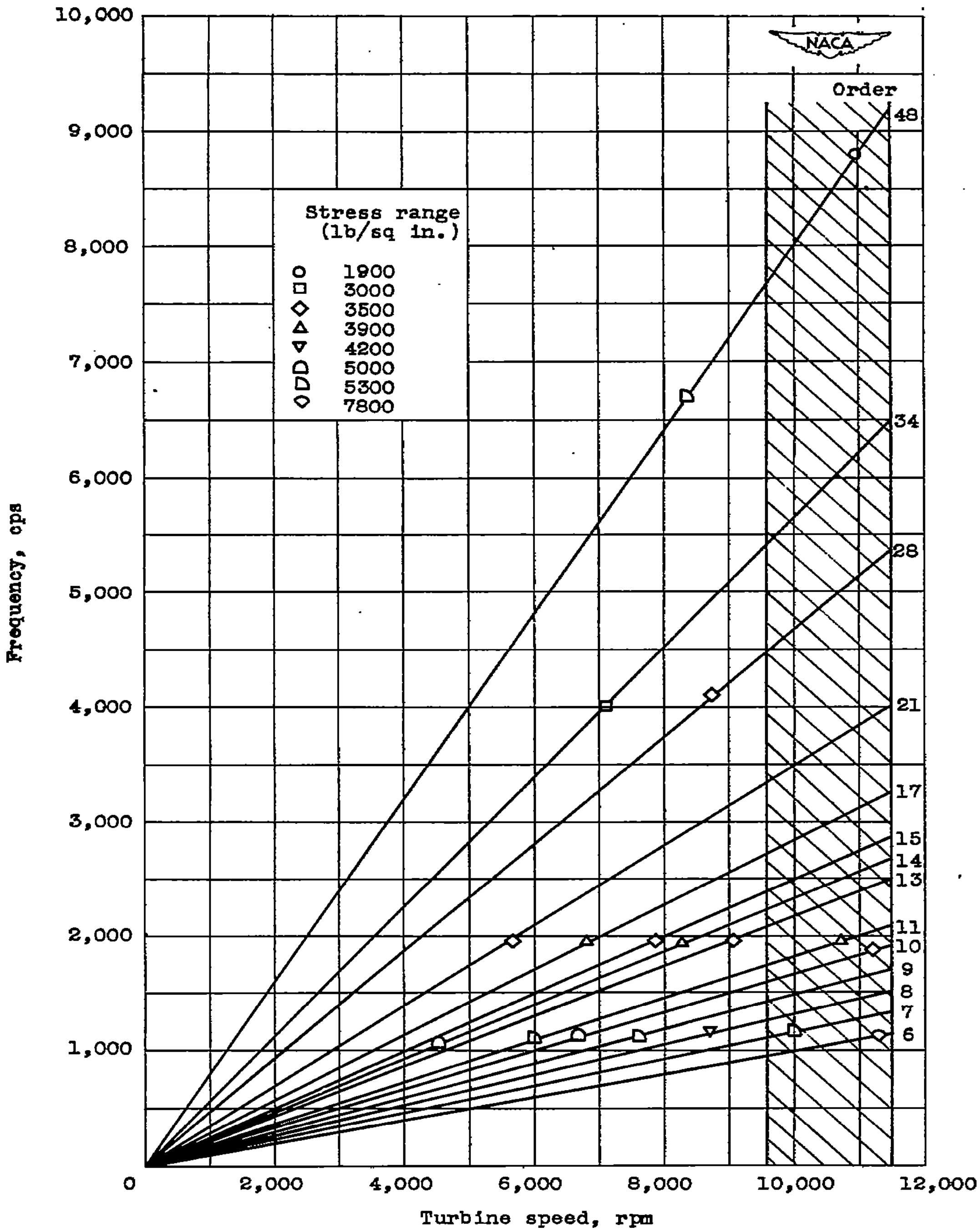

Figure 4. - Critical-speed diagram of vibration occurring in loosely mounted turbine blade with 0.03-inch-amplitude blade-tip movement. Hatched area indicates cruising range.

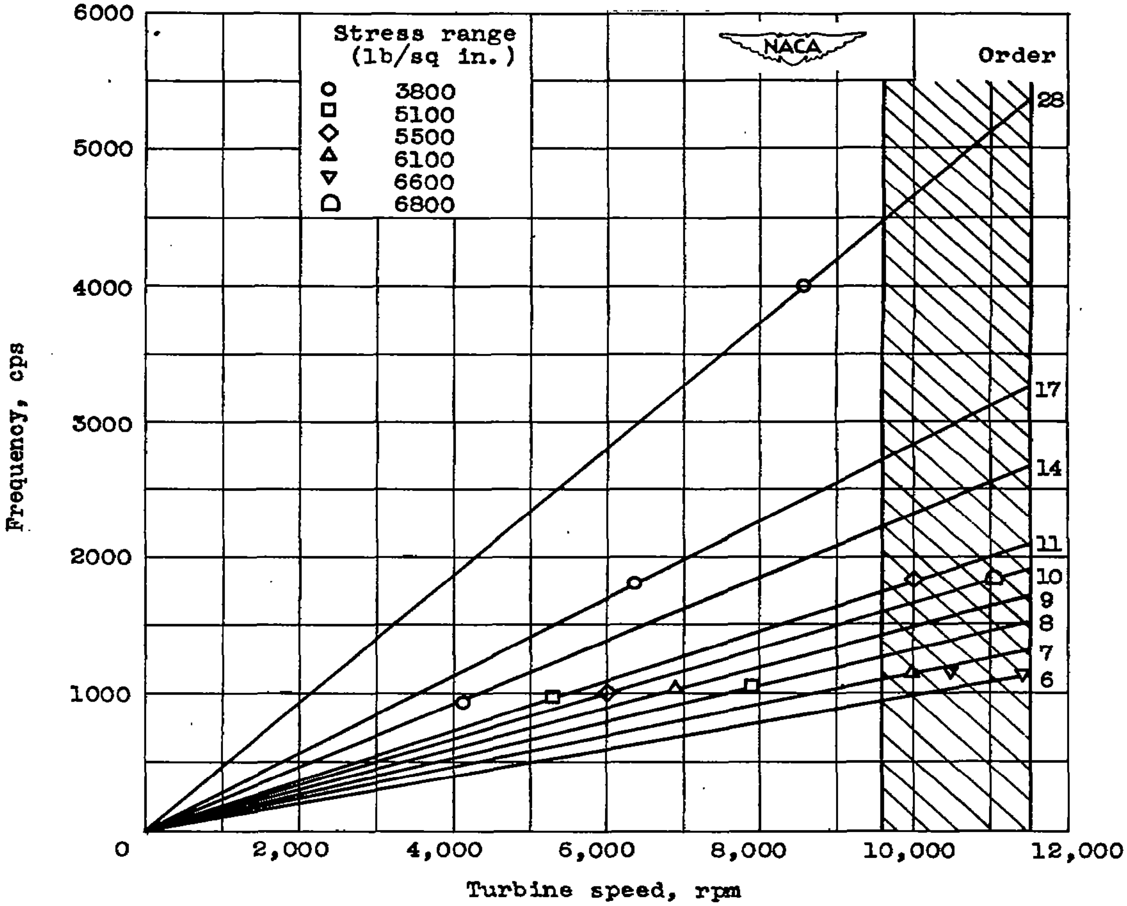

Figure 5. - Critical-speed diagram of vibration occurring in loosely mounted turbine blade with 0.06-inch-amplitude blade-tip movement. Hatched area indicates cruising range.

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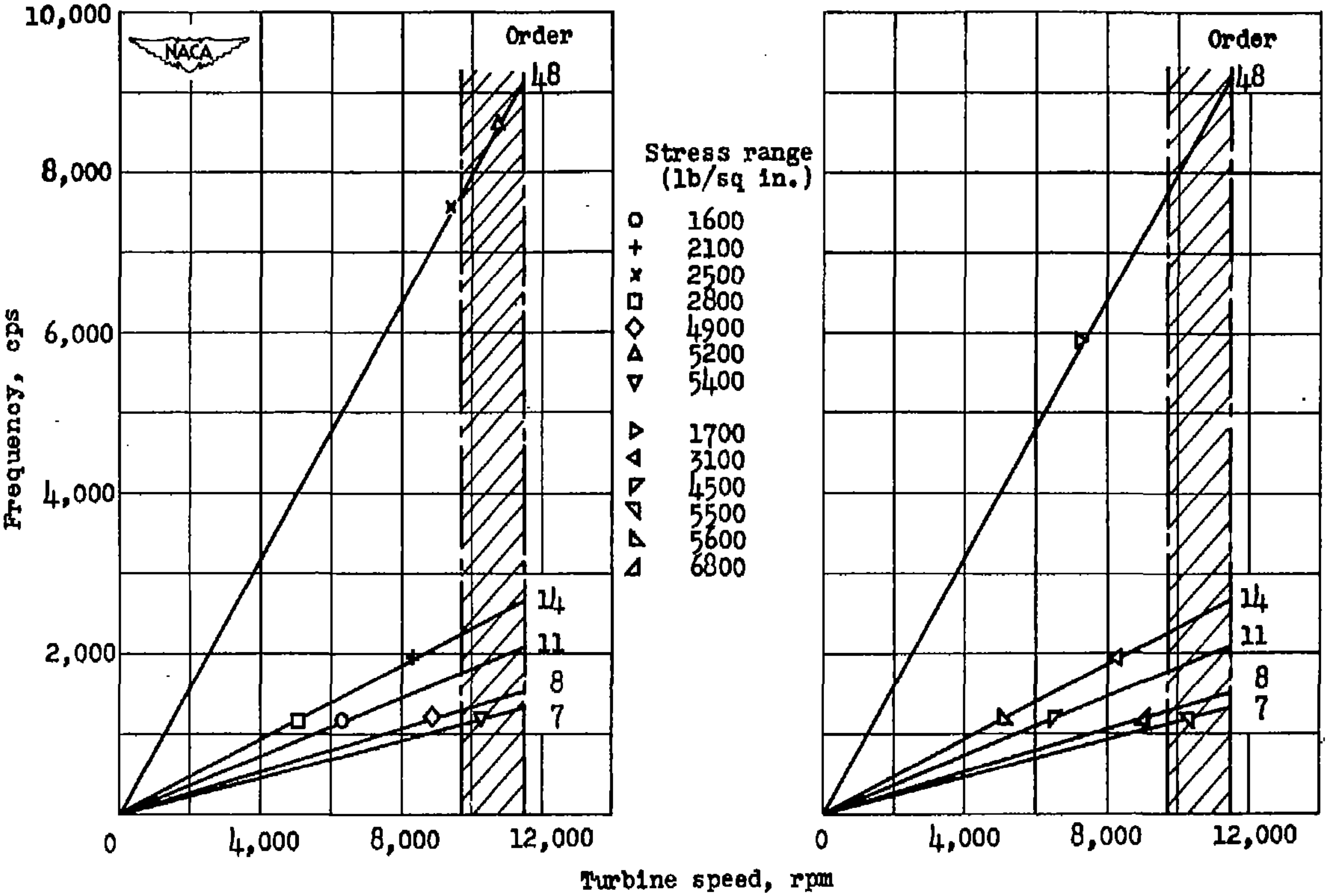

(a) Blade A.

(b) Blade B.

Figure 6. - Critical-speed diagrams of vibration occurring in tightly mounted turbine blades. Hatched area indicates cruising range. (Fig. 6 of reference 4.)

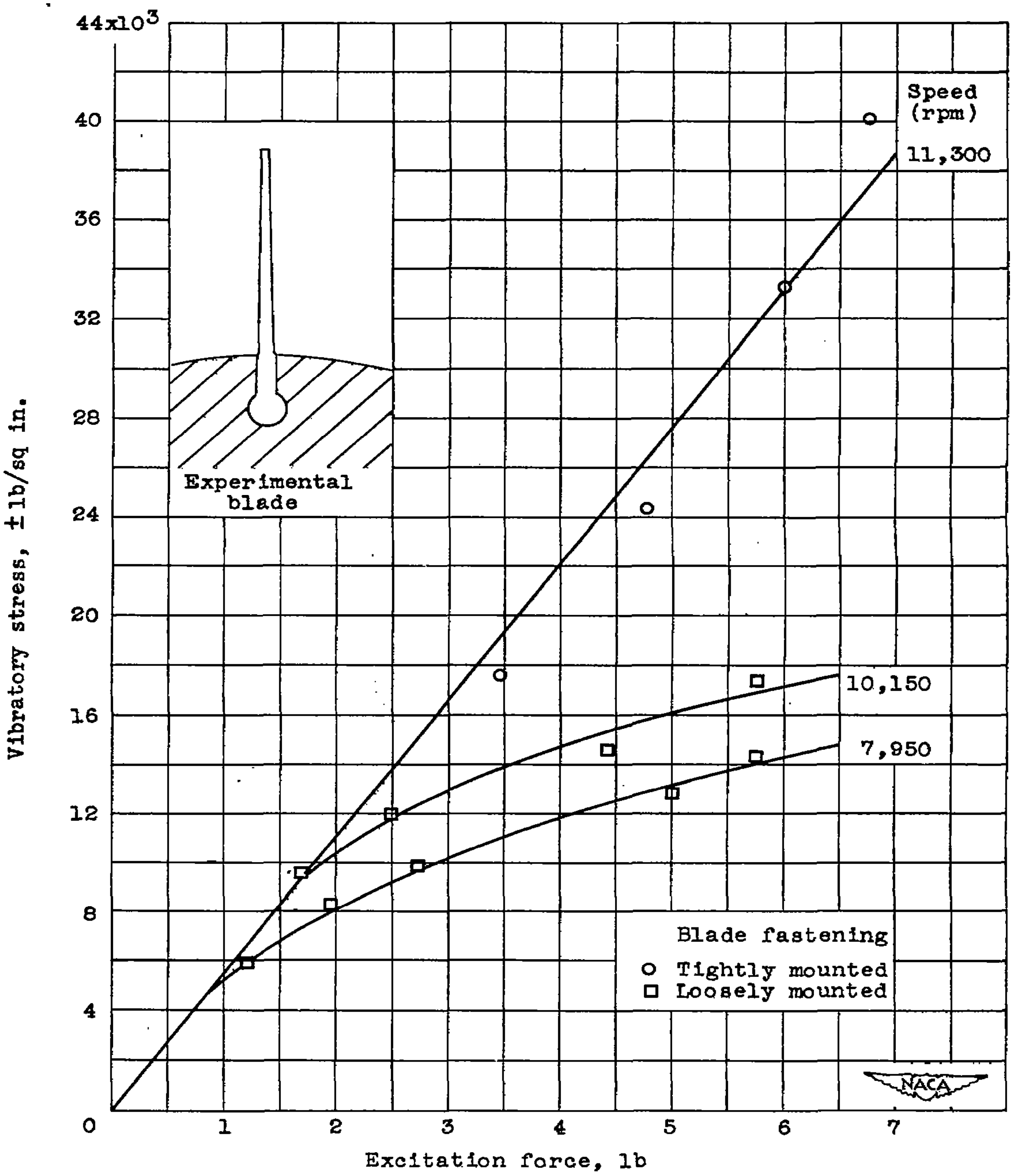

Figure 7. - Effect of vibratory stress on magnitude when blade fastening and excitation forces are varied. (Data taken from reference 5.)

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